Development of a Weather Correction Model for Outdoor Vehicle Testing

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Abstract

Development of a Weather Correction Model for Outdoor Vehicle Testing. DANIEL TUHUS-DUBROW (Brown University, Providence, RI 02912) Dr. Rom McGuffin (National Renewable Energy Laboratory, Golden, CO 80401).

When a vehicle sits all day in the sun, its cabin air temperature can reach as high as 80°C, and the dash temperature can reach 120°C. This requires a great deal of air-conditioning power for the initial cool-down of the vehicle. The National Renewable Energy Laboratory is currently looking into methods for reducing peak solar loads in vehicles, examining such technologies as solar reflective glazings, improved thermal insulation, and ambient venting systems. Two Lincoln Navigators are being tested outside the Thermal Test Facility for this purpose. One problem with outdoor testing of any kind is that the weather is always changing, and this could have an important effect on the results of the test. For example, if one technology was tested on a cloudy day, and another one on a sunny day, comparing the results would be meaningless. In order to account for these variations, a weather correction model has been developed. This is a two-node model that predicts the temperature rise in the cabin air and the cabin mass. "Standard" weather conditions are then chosen, and the measured data are normalized to these standard conditions so that different tests can be meaningfully compared. Results from the model are promising, but more testing must be done before the weather corrector can be put into use.

ERULF -- Engineering

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Introduction

While sitting all day beneath the summer sun, a vehicle's cabin air temperature can reach as high as 80°C, and the dash temperature can reach 120°C. Currently, air conditioning (a/c) systems are designed to cool off the inside cabin air as quickly as possible, without consideration of the impact on fuel efficiency. Because of this, a/c systems are greatly overpowered, typically requiring approximately 4000 W, while the human body only dissipates about 100 W. For a vehicle that normally attains 40 mpg without any auxiliary loads, a 1000 W load reduces the fuel economy by 5 mpg (Farrington et. al 1998). The effects of reducing these vehicle auxiliary loads, which consist mainly of air-conditioning, are immense. A universal 1 mpg increase in fuel economy would save over $4 billion per year nationally, and a 5% reduction in fuel consumption would save over $5 billion per year and 127 million barrels of oil per year. (Farrington et. al 1999).

The National Renewable Energy Laboratory (NREL) is currently looking into ways to reduce peak thermal loads in vehicles. Different technologies being developed include solar reflective glazings, window shades, improved thermal insulation in the vehicle cabin body, and venting systems utilizing ambient air. The goal of this research is to ensure that peak cabin air temperatures never rise above 58°C and that the dash never gets above 68°C. Two Lincoln Navigators are being tested for these purposes outside of the Thermal Test Facility (TTF).

One problem with conducting outdoor tests of any kind over a long period of time is that the weather is always changing, and therefore will have an important effect on the results. For example, different results will be obtained when it is cloudy rather than sunny, or when there are extremely strong winds. If one technology was tested on a cloudy day, and another one on a sunny day, comparing the results would be meaningless. Furthermore, tests conducted in different locations will be subject to different conditions, as will tests conducted at the same location at different times throughout the year. In actuality, variations in weather do not usually play a factor in vehicle tests. This is because automobile manufacturers perform the tests in Phoenix, where the weather does not vary much, over a duration of only a few days. NREL, however, is conducting tests throughout the entire summer in Denver, where the weather is much more erratic than in Phoenix.

In order to account for these variations, a weather correction program has been developed. This program models the temperature rise of the cabin mass and the cabin air of the vehicles using inputs of solar radiation, wind speed and direction, ambient temperature, and barometric pressure. A set of "standard" weather conditions is then chosen, and by utilizing the corrector tests taken under different weather conditions can be meaningfully compared by normalizing the results to these standard conditions. The model's main purpose is to correct for variations in the weather, but it also has further applications. For example, it can serve as a quick way to see what the effects on thermal load would be by changing different parameters, such as the reflectivity of window glazings.

Materials and Methods

Two Lincoln Navigators recently obtained from the Ford Motor Company are being tested outside the Thermal Test Facility (TTF). The vehicles are virtually identical, coming one after the other off the assembly line. Experiments are being performed to determine the best way to keep temperatures down in the cabin. These include completely covering one vehicle with aluminum foil, covering the windows with shades, and applying solar reflective glazings to the windows. The idea is to prevent the vehicle from overheating in the first place, so that not as much air-conditioning is required for the initial cool-down of the cabin. Since tests will be performed throughout the entire summer, it is necessary to correct for the weather in order to effectively compare the different technologies.

The weather correction model has been developed in MATLAB®, a powerful mathematical computing language. A primary goal of the project was to make the model as simple as possible without sacrificing a great deal of accuracy. To this end, the model consists of just two nodes, the cabin mass and the cabin air. It would have been possible to create a model with thousands of nodes using computational fluid dynamics (CFD) software. This is not feasible, however, since the computing power involved would be inordinate. One requirement of the model is that it must examine the transient response of the vehicle, since all of the input parameters change with time. Each time step in a CFD model would take approximately ten minutes of computation time, and there are almost seven hundred time steps in the model.

Each vehicle is outfitted with 51 thermocouples, which are used to obtain a thorough temperature distribution throughout the cabin. The thermocouples measure seven air temperatures in different regions of the cabin, seven window temperatures, twenty-nine interior temperatures (seats, floor, instrument panel, etc.), and eight exterior temperatures. For the purposes of the model, the seven air temperatures are averaged into one value (representing the cabin air), as are the 44 cabin temperatures (representing the cabin mass). The ambient air temperature is also measured, so that we can see how large the temperature difference is between inside and outside the cabin, which is an important parameter.

In order to model how the temperature changes with time in the vehicle, the energy balance is employed. Simply put, this states that the change in temperature within the vehicle is proportional to the energy incident on the vehicle. A system of ordinary differential equations (ODEs) results from the application of the energy balance, which must be solved in MATLAB using a numerical ODE solver. The two equations used in the model are:

$$M_{CM} C_{CM} \frac{dT_{CM}}{dt} = A_{CM}(q''_{sun} + q''_{sky} - q''_{rad} - q''_{conv,i} - q''_{conv,o})$$
$$M_{CA} C_{CA} \frac{dT_{CA}}{dt} = A_{CA}(q''_{conv,i})$$

where the subscript *CM* refers to the cabin mass and the subscript *CA* refers to the cabin air. *M* is mass, *C* is specific heat, *T* is temperature, and *t* is time. *A* is the effective area across which heat flows, q''_{sun} is the direct solar radiation incident on the vehicle, q''_{sky} is the diffuse solar radiation, q''_{rad} is the energy re-radiated by the vehicle into the atmosphere, $q''_{conv,o}$ is the convective heat transfer from the cabin mass to the ambient outside air, and $q''_{conv,i}$ is the convective heat transfer from the cabin mass to the cabin air. Each of these terms will be examined individually in more detail.

Solar radiation is the only energy input into the vehicle, but it is quite a potent source. It is the most important factor in the model, and therefore it is critical to use accurate solar radiation data. This data is obtained from the Reference Meteorological and Irradiance System (RMIS) operating at the Outdoor Test Facility (OTF) of NREL. The OTF is only a few hundred feet from the TTF, so conditions are similar enough for the purposes of the model. Thousands of readings are taken daily and put into one minute averages for the day.

The model makes use of two solar radiation measurements obtained from RMIS -- the global normal and diffuse horizontal components. Global normal refers to the direct radiation received in a line from the sun plus the diffuse component in that direction. Diffuse horizontal refers to the diffuse radiation falling on a horizontal surface. Diffuse radiation consists of the solar radiation that is scattered in the atmosphere (see figure 1). The portion of total solar radiation that is diffuse is about 10% to 20% for clear skies and up to 100% for cloudy skies (Marion et. al).

As the sun moves across the sky, the incident radiation on different parts of the vehicle changes. In the morning, for instance, the west side of the vehicle will be shaded, while in the afternoon, it will be receiving direct solar radiation. In order to account for this, the vehicle was split into four parts – the front, two sides, and the back. The solar radiation incident on each part was determined using trigonometric relations involving the zenith and azimuth angles of the sun (see figure 2). The solar angles used were taken from a solar radiation data file for Boulder.

One of three things can happen to radiation incident on a surface – it can be reflected, absorbed, or transmitted. Since there is only one node for all of the cabin mass, both the absorbed and transmitted components of the incident solar radiation contribute to heating up the mass.

If solar radiation was the only factor operating on the vehicle, the vehicle would heat up indefinitely at a very high rate. Fortunately, there are other processes in action, such as re-radiation and convection, which carry energy away from the vehicle. As the cabin mass heats up due to solar radiation, it re-radiates energy back into the atmosphere according to the Stefan-Boltzmann Law:

$$q_{rad} = (\varepsilon_m A_m + \varepsilon_w A_w)\sigma(T_{CM}^4 - T_{ambient}^4)$$

where σ is the Stefan-Boltzmann constant, ε_m and ε_w are the emissivities of the metal and the windows respectively, A_m and A_w are the areas of the metal and windows, and $T_{ambient}$ refers to the temperature of the outside ambient air.

Wind causes convective heat transfer to occur on the cabin exterior and as a result cools off the vehicle:

$$q''_{conv,o} = h_o(T_{CM} - T_{ambient})$$

where h_o is the outside convection coefficient. The problem of convection is one of the most difficult in the area of heat transfer. It is virtually impossible to derive accurate equations from first principles. Therefore, relationships are determined empirically, and correlations are used to determine the correct amount of heat transfer. In order to model the convective heat transfer caused by the wind, the vehicle was split into six parts – the two sides, the roof, the back, the windshield, and the hood. Each was then treated as a flat plate, so that a relatively simple correlation could be used. The direction of the wind also plays a role, since it determines the value of x in the equation. We handled this by deciding that if the wind was blowing in a direction between 315° and 45° or between 135° and 225°, x would be the length of the plate in the north-south direction. Otherwise, x was considered to be the length in the east-west direction. The average convection coefficient for each plate was determined using the following equation for laminar parallel flow over a flat plate (Incropera and DeWitt, 1990):

$$\overline{Nu}_x = \frac{\overline{h}_x x}{k} = 0.664\,\mathrm{Re}_x^{1/2}\,\mathrm{Pr}^{1/3} \qquad 0.6 < \mathrm{Pr} < 50$$

where $\overline{Nu}_x$ is the average Nusselt number, $\overline{h}_x$ is the average convection coefficient, x is the characteristic length, *Re* is the Reynolds number, and *Pr* is the Prandtl number (for air $Pr \approx 0.7$). Because wind speeds were not very high, flow over the plates was assumed to be laminar. The average convection coefficient for the entire vehicle was found by taking a weighted average of the six flat plate coefficients.

The cabin air is heated almost entirely by natural convection, which occurs due to density gradients in the air caused by temperature variations. As the air comes into contact with the hot cabin mass, it heats up via conduction and becomes less dense. It then rises, and cooler air takes its place. Through this process, the cabin air is heated to a fairly uniform temperature. Modeling natural convection in an enclosure such as a vehicle cabin is very difficult, and proves to be one of the weak spots in the model. As an approximation, the cabin was split into four vertical plates, and the following equation was used (Todd and Ellis, 1982):

$$h = [Ck\alpha^{1/3}](T_{CM} - T_{CA})^{1/3}$$

where C is a constant found in a table, k is the thermal conductivity, and α is the free convection modulus. For air at 30°C, this simplifies to:

$$h = 1.55(T_{CM} - T_{CA})^{1/3}$$

The total convective heat transfer was determined by summing the heat transfer from each of the four plates.

Results

The model did an excellent job of predicting the temperature rise in the cabin mass for the baseline vehicle. It also did a good job of predicting the cabin air temperature in the baseline vehicle. The error was less than ten percent for most of these test runs. When attempting to model a vehicle covered with aluminum foil, the model did not do as well, but still managed to match the general trends of the measured data. See figures 3-6 for output results from the model. The ambient air temperature is in red, the measured temperature is in green, and the modeled temperature is in blue.

Discussion and Conclusions

Once the modeled cabin air and mass temperatures closely resemble the test results, it is possible to correct for the weather. First, a meaningful dimensionless parameter *p* was defined:

$$p = \frac{(T_i - T_{ambient})_{\text{modified}}}{(T_i - T_{ambient})_{\text{baseline}}}$$

where T_i is the temperature of either the cabin air or cabin mass. The subscript "modified" refers to whichever technology was being tested at the time (aluminum foil, window shades, reflective glazings, etc.). The smaller the value of *p*, the better the technology, since it is desirable to have the vehicle temperature as close to the ambient air temperature as possible. The next step is to define a set of "standard" weather conditions, consisting of solar radiation, wind speed and direction, ambient temperature, and pressure values for each minute of the day. These conditions are then input into the model and a value of *p* is determined for a particular technology normalized to the standard day.

$$\left(\frac{\Delta T_M}{\Delta T_B}\right)_{\text{standard day}} = \left(\frac{\Delta T_M}{\Delta T_B}\right)_{\text{measured}} \left(\frac{\left(\frac{\Delta T_m}{\Delta T_B}\right)_{\substack{\text{standard day,}\\ \text{modeled}}}}{\left(\frac{\Delta T_m}{\Delta T_B}\right)_{\substack{\text{measured,}\\ \text{modeled}}}} \right)$$

where the subscript M refers to the modified vehicle and the subscript B refers to the baseline vehicle.

There are many possible improvements that could be made to the weather correction model. One is to increase the number of nodes in the model. A three-node model (cabin air, interior mass, exterior mass) would give more accurate results, but it would also be more complicated. Conduction from the exterior to the interior would have to be taken into account, as well as view factors for radiation coming from the hot interior. Developing the model beyond three nodes would compromise the project's goals of simplicity and speed. By making the model more complex, accuracy is gained, but at the expense of these two goals.

Other improvements include incorporating the effects of relative humidity, angle dependence of window transmittance, and spectral dependence of window transmittance. Relative humidity does not make much of an impact in a dry climate like Denver, but it would make a huge difference in a place like Miami or Houston. Relative humidity is a measure of how much water vapor is in the air compared to the maximum amount of water vapor the air could hold at that temperature. As air heats up, it is able to hold more water vapor. Therefore, the absolute humidity (defined as grams of water vapor per cubic meter of air) is a function of both relative humidity and temperature. Temperature, however, is what the model predicts, and is a function of absolute humidity and other factors. In order to include humidity in the model, an iterative technique of some sort would need to be used.

As the incident angle of solar radiation on a surface increases, the transmittance of the material declines dramatically. For example, if a glass surface transmits 80% of radiation falling normal to the surface, it will transmit only 40% of radiation entering at an angle of 80° (Harkness and Mehta, 1978). Materials also transmit different wavelengths in the electromagnetic spectrum more than others. This would be important if a material transmitted most of the visible light, but reflected most of the infrared.

The weather correction model does not perfectly predict the temperature rises inside the vehicle, but that may be an inherent limitation in a model that consists of only two nodes. Currently, it does a decent job of modeling the air and mass temperatures for the baseline vehicle, but is not so accurate with the various technologies being tested. With further development, it may prove to be adequate for the level of accuracy desired in these outdoor vehicle tests.

Acknowledgements

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References

Farrington, R., Anderson, R., Blake, D., Burch, S., Cuddy, M., Keyser, M., & Rugh, J. (1999). "Challenges and Potential Solutions for Reducing Climate Control Loads in Conventional and Hybrid Electric Vehicles." Golden, CO: National Renewable Energy Laboratory, presented at the VTMS4. pp. 3.

Farrington, R., Brodt, D., Burch, S., & Keyser, M. (1998). "Opportunities to Reduce Vehicle Climate Control Loads." National Renewable Energy Laboratory, presented at the EVS-15 Conference. pp. 2.

Harkness, E., Mehta, M. (1978). Solar Radiation Control in Buildings. Applied Science Publishers Ltd, London. pp. 103.

Incropera, F., DeWitt, D. (1990). Introduction to Heat Transfer, 2nd Edition. John Wiley & Sons, New York. pp. 366.

Marion, B., Riordan, C., & Renne, D. "Shining On: A Primer on Solar Radiation Data." National Renewable Energy Laboratory. Chapter 4.

Todd, P., Ellis, H. (1982). Applied Heat Transfer. Harper & Row Publishers, New York. pp. 140.

Figures

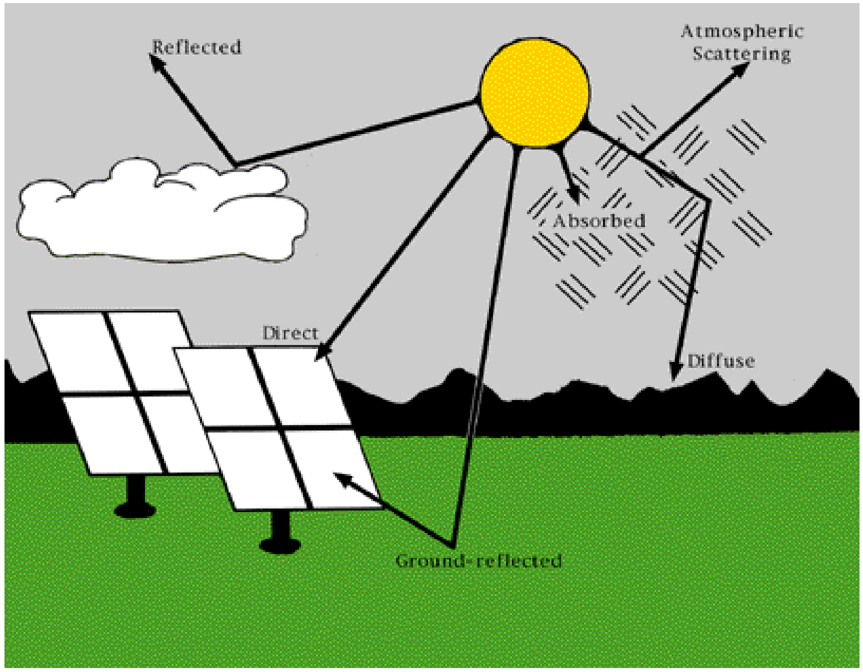

Figure 1. Shows different components of solar radiation.

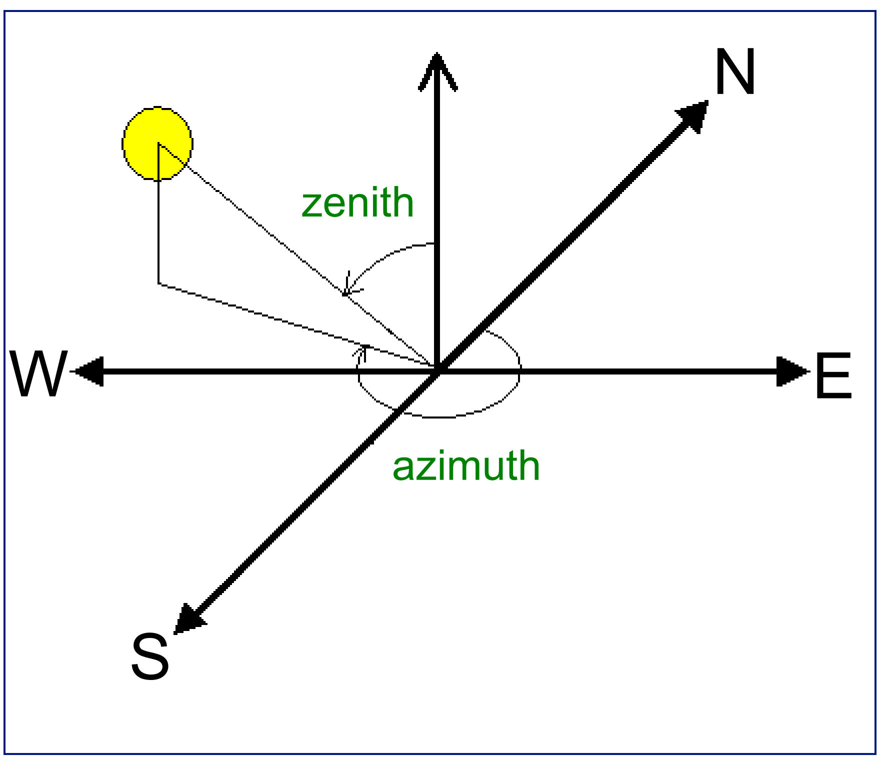

Figure 2. Shows azimuth and zenith angles.

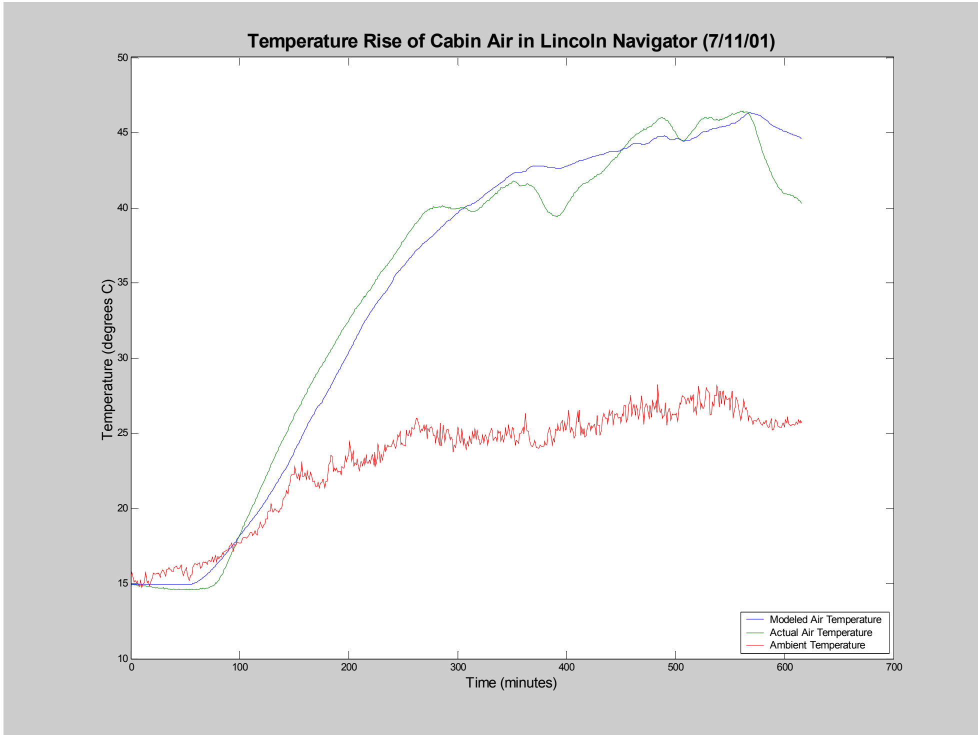

Figure 3. Shows modeled vs. actual temperature of cabin air in baseline soak.

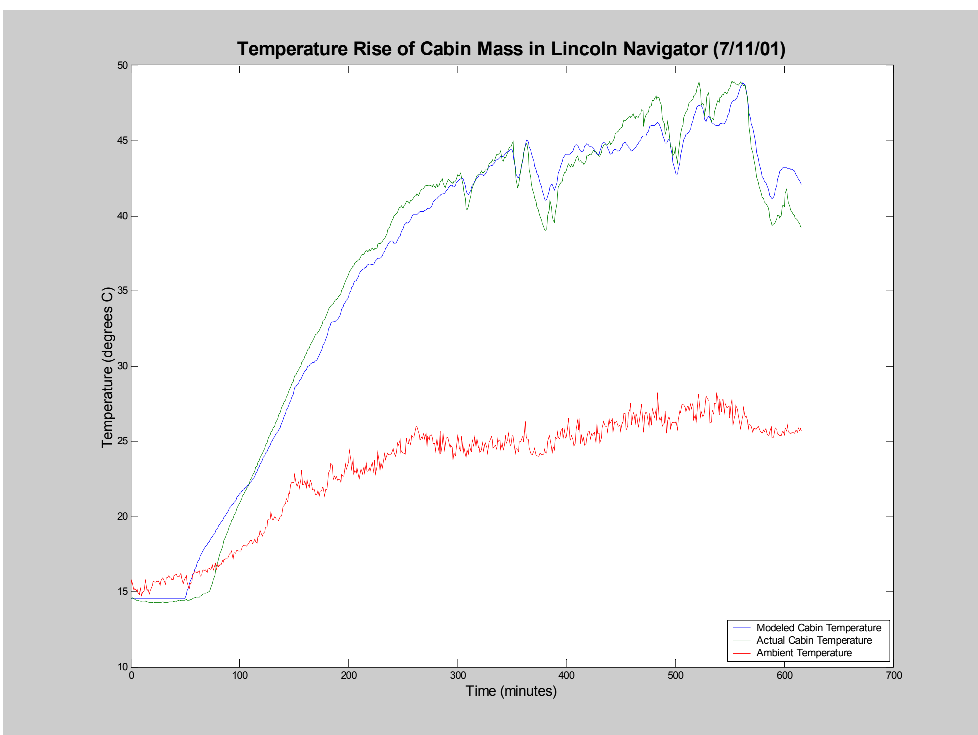

Figure 4. Shows modeled vs. actual temperature of cabin mass in baseline soak.

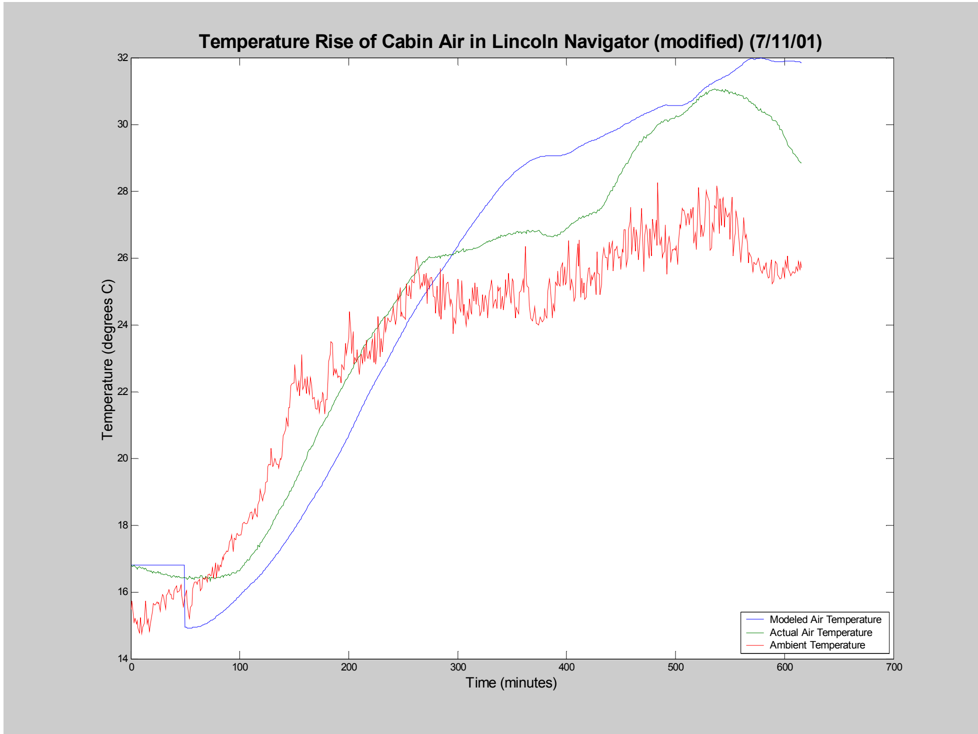

Figure 5. Shows modeled vs. actual temperature of cabin air in vehicle covered with aluminum

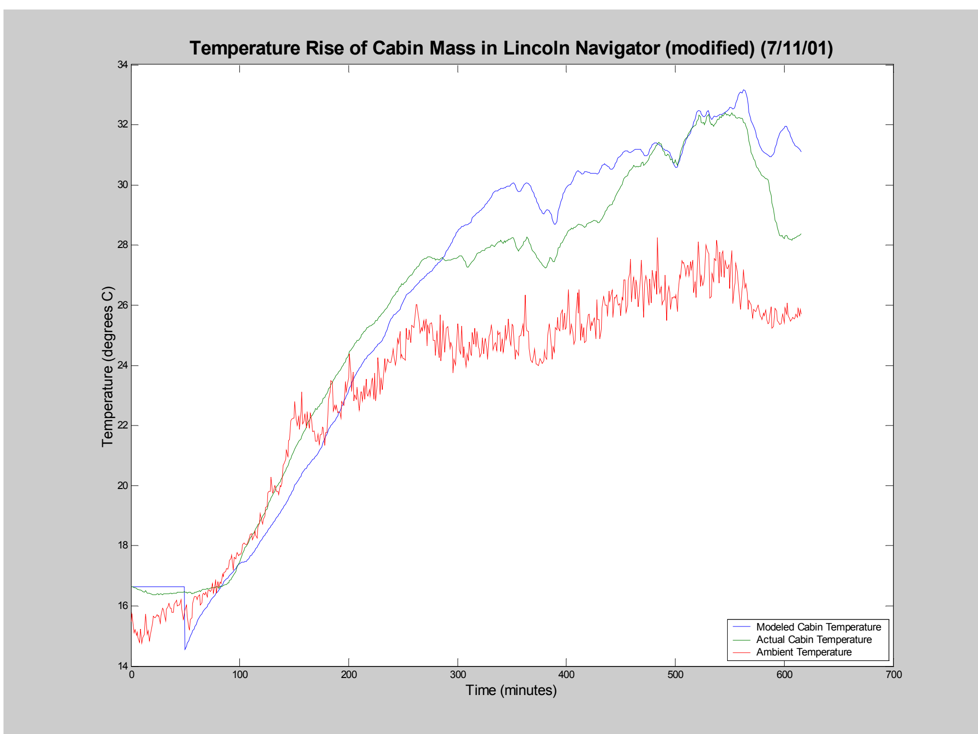

Figure 6. Shows modeled vs. actual temperature of cabin mass in vehicle covered with aluminum.